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NASA Gear Research and its Probable Effect on Rotorcraft Transmission Design

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Abstract

NASA Lewis Research Center devised a comprehensive gear technology research program beginning July 1969. The results of the NASA gear research are being integrated into the NASA civilian Helicoper Transmission System Technology Program. The paper reviews the results of this gear research and those programs which are presently being undertaken. Research programs studying pitting fatigue, gear steels and processing, life prediction methods, gear design and dynamics, elastohydrodynamic lubrication, lubrication methods and gear noise are presented. The impact of advanced gear research technology on rotorcraft transmission design is discussed.

Introduction

The requirements for advanced helicopter transmission and aircraft engine gearboxes include weight reduction, higher temperature operations than present day aircraft, as well as increased reliability and service life. The gearing in these aircraft is expected to carry greater loads, operate at higher temperatures because of increased engine speeds, provide improved system life, in addition to providing low maintenance rates and higher reliability. Elevated temperature operation of gears is also required where the transmission must operate for short periods without lubrication and cooling without resulting in a catastrophic failure.

The failure characteristics and mechanical properties must be defined in existing and potential gear materials before improvements can be made in gear material technology. Three possible approaches to improve the state-of-the-art in gear material technology can be pursued individually or simultaneously. These consist of (a) gear life testing coupled with failure analysis, (b) improving gear material properties, and/or (c) exploring new and improved gear designs.

One of the limitations of gear technology which prevents meeting more stringent aircraft requirements is a lack of knowledge relating to certain gear materials. This lack of knowledge relates to a material notch sensitivity and surface load-carrying capacity. Gear teeth will generally fail because of tooth breakage and surface distress (scoring) in addition to surface pitting (rolling-element fatigue). Increased gear-tooth loading will, of course, aggravate these problems.

Tooth bending endurance testing has been performed on gears over a period of several decades. However, the results of such tests have not been definitive [1]. Results obtained in rolling-element fatigue tests [2] show that the following parameter can significantly affect fatigue life material hardness, material heat treatment, lubricant type and batch, temperature, surface finish, operating speed, and contact stress. Unfortunately, these variables have not been carefully controlled (or have not been controlled at all) in gear testing. In some instances, insufficient tests were statistically inconclusive. Furthermore, some tooth bending fatigue tests which have been performed [3] have resulted in fretting fatigue rather than bending fatigue, wherein tooth fracture occurred at an incipient spall caused by fretting.

Design methods for the avoidance of gear tooth breakage are based on the bending endurance limit of the gear material. Usually in these methods the gear tooth is analyzed as a cantilever beam with the addition of semiempirical service and geometry factors. If the maximum calculated bending stress is less than the endurance limit strength of the material then it is presumed that no tooth breakage will occur [1,4,5].

Current methods of design to resist surface fatigue are based on the concept of a surface fatigue endurance limit. The current method [7-9] of predicting gear tooth pitting failures is similar to that used for predicting tooth breakage. However, there does not appear to be an endurance limit for this mode of failure [9-11]. Life prediction methods for rolling-element bearings are based upon the bearings having a finite life at all stress levels [12,13]. In [14] stress levels are defined for two grades of steel for which finite life is predicted at 10^{10} cycles. This approaches the method used for rolling-element bearings.

Gear lubrication and cooling become an important consideration in the design and successful operation of mechanical power transmissions. As a first step in understanding the cooling phenomena in gears, it is important to understand how oil penetrates into the gear tooth spaces under dynamic conditions. This is necessary in order to determine how much of the impinging oil is involved in the cooling and lubrication process and how much of the lubricant is "flung-off." In addition, it is important for the design engineer to be able to specify a sufficient oil jet pressure to assure adequate oil jet penetration into the root region of the gear teeth. The oil jet "impingement depth" is the point where the lubricant jet collides with the gear tooth while the "penetration depth" is the maximum depth of the lubricant penetration after impingement. The penetration depth is usually larger than the impingement depth. An analytical model [15] developed to determine these parameters was not experimentally verified and did not consider the effects of windage.

The type of lubrication, which prevents or minimizes surface asperity interaction is referred to as elastohydrodynamic lubrication. This lubrication mode is differentiated from boundary lubrication which comprises essentially metal-to-metal contact with a chemical or oxide film preventing gross wear or welding of the asperities. In most gear applications a combination of elastohydrodynamic and boundary lubrication exists. Definition of those parameters which affect the lubrication of gears must be obtained and applied to gear design and operation. These parameters include surface finish, tooth design, lubricant type, and elastohydrodynamic lubrication principles [16].

Based upon the above, the NASA Lewis Research Center devised a comprehensive gear technolgy research program beginning July 1969. The results of the NASA gear research are being integrated into the NASA civilian Helicopter Transmission System Technology Program. This paper reviews the results of this gear research [17-30] and those programs which are presently being undertaken.

Rolling-Element (Pitting) Fatigue

The metallurgical processing imposed on a gear steel from its elemental plate to the finished component can significantly affect its ultimate performance. Even the type of ore from which the various elements are extracted can exercise some influence over later component life. Theoretically then, a large number of variables could be considered in determining the rolling-element (surface pitting) life of a potential gear material. This becomes a nearly impossible task.

There is only a small body of published data on material effects on gear pitting life. Many of the gear alloy improvement programs have been evaluated by mechanical tests rather than by rolling-element component or full-scale gear surface pitting fatigue tests. Since rolling-element fatigue is a unique property, it is not, as such, necessarily possible to correlate it with more standard mechanical tests [31,32].

An extensive program is being conducted by NASA to evaluate in rolling-element fatigue current state-of-the-art as well as potential gear materials. Tests have been conducted on the NASA Lewis Research Center's gear fatigue apparatus (fig. 1) and the General Electric's rolling-contact tester (fig. 2) [33,34]. A representative fatigue (pitting) spall on a gear tooth is shown in figure 3.

RC Test Results - A summary of RC testing results are shown in figure 4. Figure 5 is a typical rolling-element fatigue spall on a RC specimen. The rolling-element fatigue life of CVM AISI 9310 is equivalent to or slightly better than VIM-VAR AISI M-50. The double vacuum process did not improve rolling-element fatigue life over single vacuum melted AISI 9310.

It was found that the rolling-element fatigue life is dependent on the level of retained austenite in the case structure. In the range of 10 to 20 volume percent retained austenite in the case, 10-percent life of AISI 9310 was shown to be increased more than three times. Little change in life was observed between specimens with 8.3 and 11.2 percent retained austenite.

The rolling-element fatigue life of CBS 600 was found to be equivalent to that of CVM AISI 9310 and VIM-VAR AISI M-50. The effects of different tampering temperatures employed and the introduction of freezing cycles on rolling-element fatigue do not appear to be significant.

The results with CBS 1000 M was found to be interior to VIM-VAR AISI M-50 and CVM. CVM Vasco X-2 was found to be equivalent to these two materials.

The rolling-element fatigue life of the nitriding alloys, VIM-VAR Super Nitralloy, air melt Nitralloy 135 and CVM Nitralloy N was found to be less than VIM-VAR AISI M-50 and to CVM AISI 9310.

Full-Scale Gear Test Results - Full-scale gear tests were performed with the materials enumerated above [17,19,22,25,35,36] in the NASA gear fatigue test rig. Gears manufactured from air melt CBS 600 exhibited lives longer than those manufactured from CVM AISI 9310. The gear manufactured from CVM modified Vasco X-2 exhibited lives statistically equivalent to CVM AISI 9310 gears. Both the CBS 600 and modified Vasco X-2 gears exhibited the potential of tooth fractures occurring at a tooth surface fatigue pit. Case curburization of all gear surfaces for the modified Vasco X-2 gears resulted in fracture at the tips of the gears.

CVM AISI M-50 gears without tip relief had lives approximately 50 percent longer than CVM Super Nitralloy gears without tip relief. However, Super Nitralloy gears with tip relief had lives equal to the CVM AISI M-50 gears without tip relief However, the difference in lives were not statistically significant. All gears failed by classical pitting fatigue at the pitch circle. However, the CVM AISI M-50 gears with tip relief failed by tooth fracture. CVM AISI M-50 gear sets without tip relief having a spalled gear tooth which were deliberately overrun after spalling had occurred, failed by tooth fracture [19].

Gear Forging

Gears made from materials which are through hardened or case carburized materials having a high percentage of alloying elements, have a tendency for gear tooth fracture due to bending fatigue after extended running subsequent to a surface fatigue spall. Figure 6 is a typical tooth fracture eminating from a surface fatigue spall. One fabrication method which has the potential to improve the strength and life of gear teeth is termed "ausforging." Ausforging is a thermomechanical metal working process whereby a steel is forged or otherwise worked while it is in the meta-stable austenitic condition [2]. A number of researchers have investigated this process [1,3-5]. The application of ausforging to machine elements such as rolling-element bearings was first reported in [37].

Tests were conducted [25,28] at 350 K (170° F) with three groups of 8.9 cm (3.5 in.) pitch diameter spur gears made of VIM-VAR AISI M-50 steel and one group of CVM AISI 9310 steel. The pitting fatigue life of the standard forged and ausforged gears was approximately five times that of the CVM AISI 9310 gears and ten times that of the bending fatigue life of the standard machined VIM-VAR AISI M-50 gears run under identical conditions. There was a slight decrease in the 10-percent life of the ausforged gears from that for the standard forged gears. However, the difference is not statistically significant.

The standard machined gears failed primarily by gear tooth fracture while the forged and ausforged VIM-VAR AISI M-50 and CVM AISI 9310 gears failed primarily by surface pitting fatigue. The ausforged gears had a slightly greater tendency to fail by tooth fracture than the standard forged gears.

While gear forging offers the potential for long-lived reliable gearing, especially for the high alloy steels, both the cost and the availability of forging facilities may outweigh its advantages at the present time. It may, however, be necessary to consider forging gears for such materials as CBS 600 and modified Vasco X-2 where there is a potential for tooth fracture eminating from surface fatigue spalls.

Gear Life Predictions

The fatigue-life model proposed in 1947 by Lundberg [12] is the commonly accepted theory to determine the fatigue life of rolling-element bearings. The probability of survival is expressed as follows:

$$\log \frac{1}{S} \propto \frac{\tau_0^c \eta^e}{z_0^h} V \tag{1}$$

where

S probability of survival

V volume representation of the stress concentration or "stressed volume"

η millions of stress cycles

e Weibull slope

h,c material dependent exponents

τ_0 critical stress

z_0 depth of the critical stress

Unfortunately no constant or porportionality was given by Lundberg and Palmgren for equation (1). However, by working back from a material constant given near the end of their paper the constant used in equation (2) was determined [22]. Therefore, the equation for life with a 90-percent probability of survival may be written as follows:

$$L_1 = \left(\frac{K_1 z_0^h}{\tau_0^c V}\right)^{1/e} \tag{2}$$

where

$$K_1 = 1.430\times10^{95} \quad \text{(SI units)}$$

$$= 3.583\times10^{56} \quad \text{(English units)}$$

This constant was found to be valid for common bearing steel of 1950 vintage (AISI 52100)[13].

Based on life tests for roller bearings the accepted values for the exponents are $h = 2\frac{1}{3}$, $c = 10\frac{1}{3}$, $e = 1\frac{1}{2}$.

In the Lundberg and Palmgren theory, the load-life exponent for line contact is $p = (c - h + 1)/2e$. The Lundberg-Palmgren e and p are primary exponents which were obtained from bearing tests. The values of c and h were obtained from e and p and the results of tests made with a series of different sized bearings. The values of h and c are accepted for use in this paper, but the value of $e = 3$, which is based on gear tests reported in [14,15,23] will be used in the calculation for gear life. Based on these values of h, c, and e a value of $p = 1.5$ results.

Much of the work by Lundberg and Palmgren was concerned with connecting the basic equation to common bearing geometry and operating parameters. In order for the theory to be directly useful and not involve cumbersome calculations, the same approach was used by NASA [23,24] for gears. In Table 1 a rational way of treating the stress, stressed volume, and number of stress cycles for gear systems is presented.

The Weibull slope e and the load-life exponent p may be directly determined by conducting life tests under several load conditions for a group of gears. Life tests were conducted in three different gear loads with three groups of AISI 9310 gears [30]. The results of these tests are shown in figure 7. Life was found to vary inversely with load to the 4.3 and 5.1 power at the 10-percent and 50-percent life levels, respectively.

Using the procedure of Table 1, and the values determined experimentally from the gear tests, the life distribution for the three groups of AISI 9310 gears were calculated. These distributions are plotted for comparison with the experimental data in figure 8.

The American Gear Manufacturers Association (AGMA) has published two standards for tooth surface fatigue [14,38]. These standards are AGMA 210.02 and AGMA 411.02. AGMA 210.02 provides for an endurance limit for surface fatigue below which it is implied that no failure should occur. In

practice, there is a finite surface fatigue life at all loads. AGMA 411.02 recognizes this finite life condition. Therefore, it does not contain an endurance limit in the load-life curve but does show a continuous decrease in life with increasing load. Both AGMA standards are illustrated in figure 9. The AGMA load-life curves shown are for a 99-percent probability of survival or the L_1 life [27]. The experimental L_1, L_{10}, and L_{50} lives are plotted for comparison.

It is evident that the load-life relation used by AGMA is different than the experimental results reported herein. The difference between the AGMA life prediction and the experimental lives could be the result of differences in stressed volume. The AGMA standard does not consider the effects of stressed volume which may be considerably different than that of the test gears used herein. The larger the volume of material stressed the greater the probability of failure or the lower the life of a particular gear set. Therefore, changing the size or contact radius of a gear set, even though the same contact stress is maintained would have an effect on gear life.

Tooth Profile and Pressure Angle

A majority of current aircraft and helicopter transmissions have a spur-gear contact ratio (average number of teeth in contact) of less than 2. The contact ratios are usually from 1.3 to 1.8, so the number of teeth in engagement is either one or two. Many gear designs use a pressure angle of 25° for improved tooth strength, giving a contact ratio of approximately 1.3. This low contact ratio causes increased dynamic loading of the gear teeth and increase noise, sometimes causing lower pitting fatigue life.

High-contact-ratio gears (contact ratio greater than 2) have load sharing between two or three teeth during engagement and, therefore, usually have less load per tooth. These gears should operate with lower dynamic loads and thus less noise.

High-contact-ratio gears have been in existence for many years but have not been widely used. High contact ratios can be obtained in several ways: (a) by smaller teeth (large pitch), (b) by smaller pressure angle, and (c) by increased addendum. As a result, high-contact-ratio gears tend to have lower bending strength and increased tooth sliding. Because of the increased sliding, the high-contact-ratio gears may run hotter and have a greater tendency for surface-distress-related failures such as micropitting and scoring.

Profile modification (changing the involute profile at the addendum or dedemdum or both) is normally done on all gears to reduce tip loading and scoring [39]. However, if it is done improperly, it could increase the dynamic load [19]. Several profile modifications have been proposed that would reduce scoring and improve the performance of high-contact-ratio gears. One such proposal is the so-called new-tooth-form (NTF) gear, which has a large profile modification at both the addendum and dedendum. The profile radius of curvature is also reduced at the addendum and increased at the dedendum in an attempt to lessen sliding and thereby reduce scoring of HCR gears. However, a gear geometry analysis [40] indicates that sliding is independent of the profile radius of curvature.

Under NASA contract, the Boeing Vertol Co. designed and manufactured two sets of NTF gears as well as two sets of standard gears for the purpose of evaluating the NTF gears and comparing them with standard gears.

Scoring tests, surface fatigue tests, and single-tooth bending fatigue tests were conducted using four sets of spur gears of standard design and three sets of spur gears of new-tooth-form design [41]. The scoring tests were conducted on a Wright Air Development Division (WADD) gear test rig at a speed of 10 000 rpm. The surface fatigue tests were conducted on the same test rig at a speed of 10 000 rpm and at maximum Hertz stresses of 173×10^7 and 148×10^7 N/m^2 (250 000 and 214 000 psi). The single-tooth bending fatigue tests were conducted on both the standard and new-tooth-form (NTF) gears starting at a bending stress of 104×10^7 N/m^2 (150 000 psi). The stress was increased until failure occurred at 3×10^6 cycles or less.

Both the standard and NTF gears scored at a gear bulk temperature of approximately 409 K (277° F). At this temperature the load on the NTF gears was 22 percent less than the load on the standard gears. The scoring failure was a function of gear bulk temperature, where for a given lubricant the temperature is a function of gear design, operating load, and speed.

The results of the surface fatigue tests are shown in figure 10. The pitting fatigue lives of the standard and NTF gears were statistically equal for the same maximum Hertz stress. The pitting fatigue life of the NTF gears was approximately five times that of the standard gears at equal torque or load.

The minimum load to produce a bending fatigue failure at 3×10^6 cycles for the standard gear tooth was 1.9 times that for the NTF gear tooth. The standard gear tooth failed at a 17-percent higher bending stress than the NTF gear tooth when stress was calculated by the American Gear Manufacturers Association (AGMA) method. However, this difference is not statistically significant.

Spur Gear Dynamic Analysis

A gear dynamic analysis has been developed for standard and high-contact ratio gears by Hamilton Standard division of United Technologies under contract to NASA. The program will predict the gear dynamic loads for standard and high-contact ratio gears with variations in gear tooth profile modifications, in addition to variations in system mass and damping. The program is currently being extended to include rim effects, interval gears and multiply gear meshes.

NASA is also conducting a spur gear dynamic

analysis program with Cleveland State University. This program is using a different approach for the dynamic analysis and has a finite element program for gear tooth deflections. Using this computer program for gear design will give a much better determination of the effects of various gear and system operating parameters on gear dynamic loads and life, and make it possible to improve the load-carrying capacity and life of gear systems.

Spiral Bevel Gears

NASA is conducting a fundamental study of spiral bevel gear technology. From a study of the current state-of-the-art in spiral bevel gear technology, significant advances can be achieved in reducing gear noise and vibration, better estimates of gear strength and life, as well as better lubrication techniques which will reduce wear and minimize temperature rise.

Based on test work with the NASA 500-hp transmission test stand it has been found that spiral bevel gear noise is significantly higher than for the spur planetory set in current helicopter designs. If gear noise in helicopter drives is to be significantly reduced the largest source must be reduced in order to make any real reductions. Also, it has been found that stresses in the root of the spiral bevel gears are significantly higher than handbook formulae would indicate.

In order to adequately understand the effects of spiral bevel gear design parameters on noise, vibration, stress, temperatures, and lubrication, the gear tooth surface geometry must be defined. Analysis is being performed to define the tooth surface geometry using differential geometry theory. Methods of optimizing the tooth surface contact are being studied.

A spiral bevel gearset computer program is being developed which makes use of mesh stiffness calculations based on finite element methods, gear shaft and bearing stiffnesses, current theories of tooth contact analysis (TCA), and elastohydrodynamic lubrication theory. The program will enable the calculation of such things as dynamic loads, bulk temperature and flash temperature, contact patterns, and lubricant film thickness. The NASA method of gear life prediction will be used as a subroutine of the gear program.

Gear Noise

Currently, under contract with Bolt, Beranek, and Newman, Inc., the gear noise problem is being studied analytically. Since noise adsorption material in helicopters is adverse to payload capacity a method of minimizing noise through better gear design is needed. This way gear noise can be minimized at the source which is at the gear mesh. The main approach is to develop a method for synthesizing optimum modifications of perfect involute tooth surfaces that will minimize the dynamic loading and noise generated by gear teeth. The work includes developing computer programs for the tooth modification synthesis procedure and for predicting the Fourier series coefficients of vibratory excitation caused by elastic tooth deformations and deviations of tooth faces from perfect involute surfaces. A gear dynamic analysis program also will be written. These programs will be used to design an optimum gear pair to be tested in-house at NASA, and to compute the expected dynamic improvement of the new design in comparison with conventional profile modifications for helicopter transmissions.

Gear Lubrication

As a first step in understanding the cooling phenomena in gears, it is important to understand how oil penetrates into the gear tooth spaces under dynamic conditions. Lubricant jet flow impingement and penetration depth into a gear tooth space were measured at 4920 and 2560 using a 8.89-cm- (3.5-in.-) pitch diameter 8 pitch spur gear at oil pressures from 7×10^4 to 41×10^4 N/m^2 (10 to 60 psi) [20]. A high-speed motion picture camera was used with xenon and high-speed stroboscopic lights to slow down and stop the motion of the oil jet so that the impingement depth could be determined (fig. 11). An analytical model was developed for the vectorial impingement depth and for the impingement depth with tooth space windage effects included. A comparison between the calculated and experimental impingement depth versus oil jet pressure is shown in figure 12 The windage effects on the oil jet were small for oil drop size greater than 0.0076 cm (0.003 in.). The analytical impingement depth compared favorably with experimental results above an oil jet pressure of 7×10^4 N/m^2 (10 psi). Some of this oil jet penetrates further into the tooth space after impingement. Much of this post impingement oil is thrown out of the tooth space without further contacting the gear teeth.

An analysis was conducted for oil jet lubrication on the disengaging side of a gear mesh. Results of this analysis is shown in figure 13. The analysis was computerized and used to determine the oil jet impingement depth for several gear ratios and oil jet to pitch line velocity ratios. An experimental program was conducted on the NASA gear test rig using high-speed photography to experimentally determine the oil jet impingement depth on the disengaging side of mesh. Impingement depth reaches a maximum at gear ratio near 1.5 where chopping by the leading gear tooth limits the impingement depth. The pinion impingement depth is zero above a gear ratio of 1.172 for a jet velocity to pitch time velocity ratio of 1.0 and is similar for other velocity ratios. The impingement depth for gear and pinion are equal and approximately one-half the maximum at a gear ratio of 1.0. Impingement depth on either the gear or pinion may be improved by relocation of the jet from the pitch line or by changing the jet angle. Results of the analysis were verified by experimental results using the high-speed camera and a well-lighted oil jet.

A computer program was developed for calculating spur gear performance characteristics [42]. The computer program consists of an iterative solution of the bulk temperature, flash temperature, local traction, and the lubricant film thickness along the path of contact. The dynamic load is

calculated from a torsional vibration analysis of the gear train. The bulk temperature is calculated from heat-transfer influence coefficients obtained from a finite-element analysis. This is solved iteratively with the elastohydrodynamic lubrication problem for the gears. It is assumed that the vibration problem is uncoupled from the thermal and EHD problem.

Typical results for dynamic load, bulk temperature, flash temperature, and EHD film thickness are shown in figure 14. The calculations were done for the case of two 20 degree pressure angle, 36T, gears running at 7800 rpm and transmitting 750 hp. Figure 14 is a computer generated plot, giving the results as a function of distance along the path of contact. The origin is taken at the pitch point.

Rotorcraft Transmission Design Impact

Current design methodology for transmission systems uses relatively standard stress calculations and methods dictated by AGMA standards. These methods for the most part have proven satisfactory for current state-of-the-art applications. However, methods for transmission life predictions are inadequate and, for the most part, inconsistent from one design group to another. The analysis being developed for life prediction will allow for consistent and more accurate life prediction methods. Transmissions can be sized not only for stress, speed, and ratio but also for life. Assuming an infinite gear pitting life is not technically acceptable. Further, comparisons from one design to another can be performed on a more objective basis.

Because of the requirement for higher temperature transmission applications new materials are being used for gears with a minimal data base and limited experience. In addition, heat treat specification and control can significantly affect the life and reliability of a gear system. Experimental definition of the relative life of gear materials and their heat treatment can aide in the selection of potential gear materials and in determining life adjustment factors for life prediction methods. Materials can rationally be selected for longer life application.

Proper gear lubrication and cooling has been basically an art. In general, lubricant flow, pressure, nozzle position and type of lubricant has been based upon prior experience and trial and error methodology. Equations have been developed whereby the position of the oil jets and lubricant flow rate can be determined with reasonable certainty to obtain efficient lubrication. Oil volume and gear operating temperatures can be optimized for a given transmission design.

The application of elastohydrodynamic (EHD) lubrication analysis to gear design and operation will enhance gear life and operation. The effect of EHD film thickness in determining transmission life and reliability has, for the most part, been ignored by the transmission designer. Furthermore, lubricant temperature impacts film thickness and gear tooth temperature. Hence, cooling analysis of the gear must be integrated with the EHD analysis. Selection of a lubricant impacts both parameters and can affect the efficiency of a gear box as well as its life.

Analytical methods to accurately predict transmission noise is another requirement which becomes a potential design tool. While life and reliability as well as efficiency are prime design considerations, alternate designs of equal merit may result in significant noise amplitudes. Proven analytical methods for predicting noise would aide in the proper design, selection or modification of gear systems whereby noise can be minimized while mechanical performance can be optimized.

Accurate dynamic stress predictions for bending and in the contact zone of gear teeth are very important factors in transmission design. Finite-element techniques for both spur and bevel gears will greatly improve current stress predictions. This would allow the design to more fully utilize the potential load capacity of advanced transmission systems and at the same time minimize transmission weight.

Tooth profile modifications and high-contact ratio gearing offer the potential to impart higher loads into a transmission for a given transmission weight. Conversely, for a given transmission load, reliability and life should improve.

In essence, the impact of the NASA gear program will as an end objective contribute technology towards more efficient transmission systems having higher power-to-weight ratios at lives and reliabilities greater than current state-of-the-art systems. Furthermore, the use of improved gear materials and design methodology should improve transmission maintainability and mean time between removal (MTBR).

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TABLE 1. - SAMPLE CALCULATION

Symbol	Description	Formula		Result	
ϕ_t	Transverse pressure angle, deg				
ψ_b	Base helix angle, deg			20	
P	Diametral pitch, teeth/cm (teeth/in.)			0	
a	Addendum			8	
N_1	Number of pinion teeth			0.318 (0.125)	
N_2	Number of gear teeth			28	
W_t	Transmitted tangential load, N (lb)			28	
	Speed of pinion, rpm			1617 (363)	
f	Face width in contact, cm (in.)			10 000	
r_1	Pinion pitch radius, cm (in.)			0.28 (0.11)	
r_2	Gear pitch radius, cm (in.)	$N_1/2P$		4.445 (1.750)	
r_{a1}	Pinion addendum radius, cm (in.)	$N_2/2P$		4.445 (1.750)	
r_{a2}	Gear addendum radius, cm (in.)	$r_1 + a$		4.763 (1.875)	
r_{b1}	Pinion base circle radius, cm (in.)	$r_2 + a$		4.763 (1.875)	
r_{b2}	Gear base circle radius, cm (in.)	$r_1 \cos \phi_t$		4.1769 (1.6445)	
p_b	Base pitch, cm (in.)	$r_2 \cos \phi_t$		4.1769 (1.6445)	
		$2\pi r_{b1}/N_1$		0.9373 (0.3690)	
ζ	Contact path length, cm (in.)	$\sqrt{r_{a1}^2 - r_{b1}^2} + \sqrt{r_{a2}^2 - r_{b2}^2} - (r_1 + r_2)\sin \phi_t$		1.5350 (0.6043)	
	Transverse contact ratio	ζ/p_b		1.64	
β_{H1}	Roll angle through heavy load zone, rad	$(2p_b - \zeta)/r_{b1}$		0.0813	
β_{L1}	Roll angle through light load zone, rad	$(\zeta - p_b)/r_{b1}$		0.1431	
δ_1	Precontact roll angle, rad	$\left[(r_1 + r_2)\sin \phi_t - \sqrt{r_{a2}^2 - r_{b2}^2}\right]/r_{b1}$		0.1802	
t_c	Minimum face width in contact, cm (in.)	$t_c - f$		0.28 (0.11)	
$\Sigma\rho$	Curvature sum, cm^{-1} (in.$^{-1}$)	$(1/r_1 + 1/r_2)\cos \psi_b/\sin \phi_t$		1.316 (3.342)	
		Case I	Case II	Case I	Case II
θ_{L1}	Roll angle when load starts, rad	$\delta_1 + \beta_{L1}$	δ_1	0.3233	0.1802
θ_{U1}	Roll angle when load ends, rad	$\theta_{L1} + \beta_{H1}$	$\theta_{L1} + 2\beta_{L1} + \beta_{H1}$	0.4046	0.5477
ℓ	Length of stressed portion of involute, cm (in.)	$r_{b1}(\theta_{U1}^2 - \theta_{L1}^2)/2$		0.1237 (0.0487)	0.5587 (0.2200)
W_{tM}	Dynamic capacity of the mesh, N (lb)	$K_2 \ell_c \cos \phi_t (\cos \psi_b)^{11/9} (\Sigma\rho)^{-35/27} \left[f \ell N_1 \left\{1 + \left(\frac{N_1}{N_2}\right)^3\right\}\right]^{-2/9}$		1[illegible] 600 (3732)	11 900 (2669)
L_M	Life of gear mesh, mr (hr)	$\left(\frac{W_{tM}}{W_t}\right)^{3/2}$		3[illegible] (54.9)	19.9 (33.2)

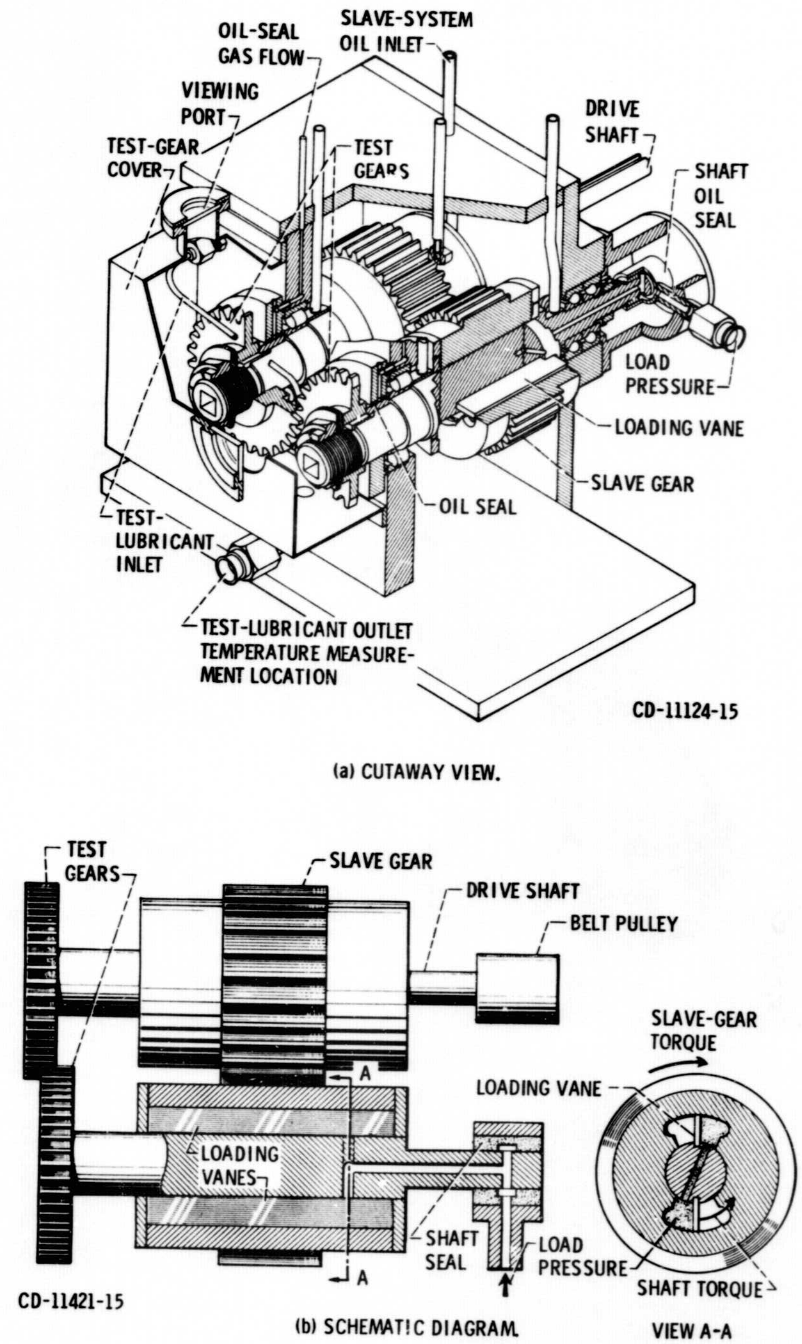


Figure 1. - NASA Lewis Research Center's gear fatigue test apparatus.

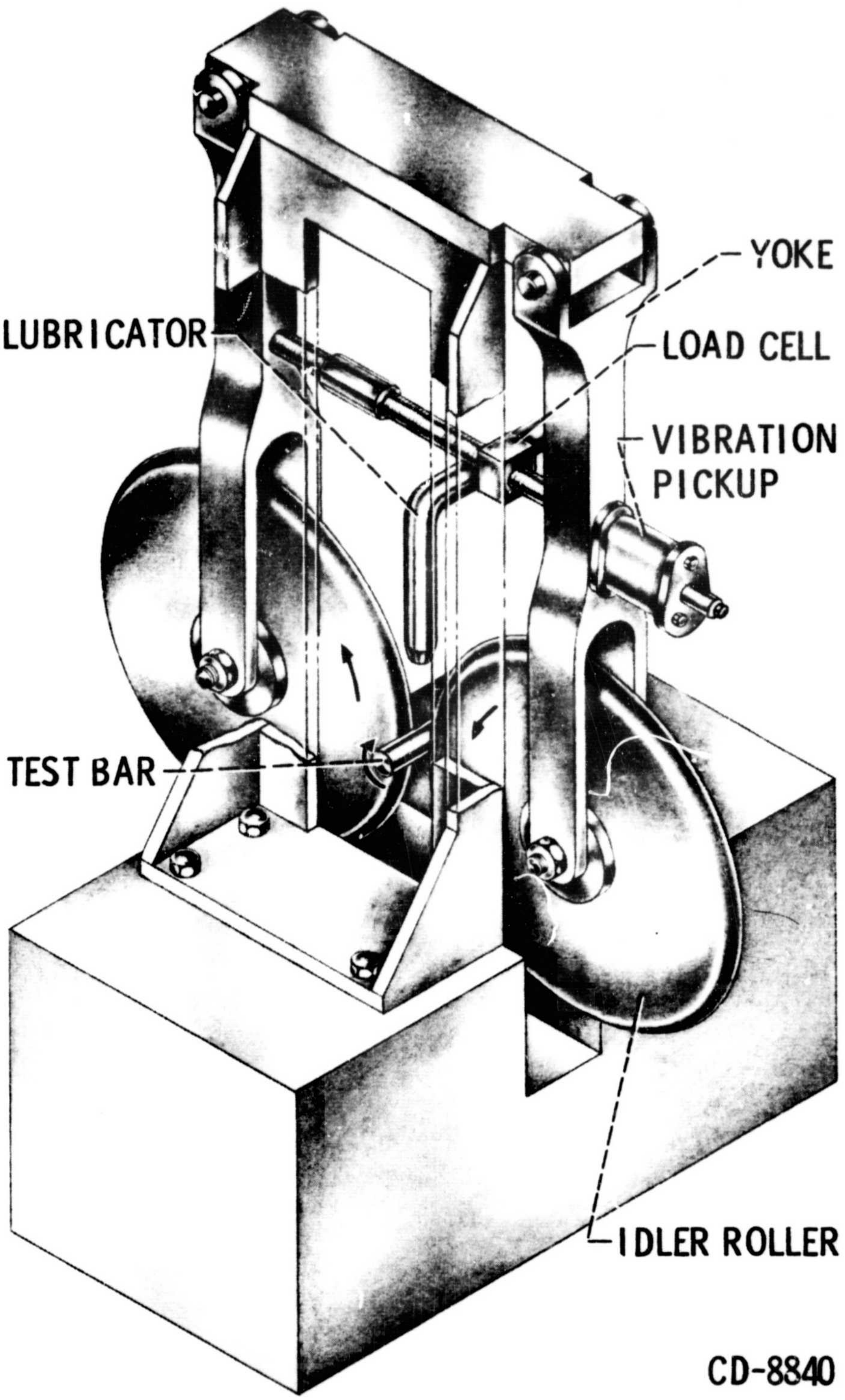


Figure 2. - Rolling-contact fatigue apparatus.

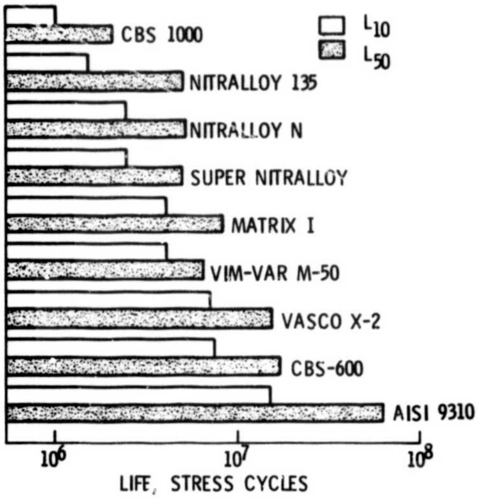


Figure 3. - Comparison of fatigue life in terms of L_{10} and L_{50} for candidate gear materials.

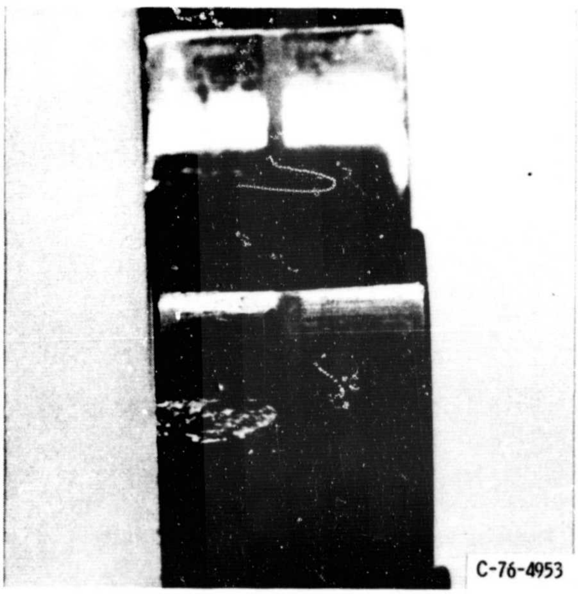


Figure 4. - Representative fatigue spall of test gear material CVM AISI 9310 steel. Speed 10 000 rpm; lubricant, superrefined naphthenic mineral oil with additive package.

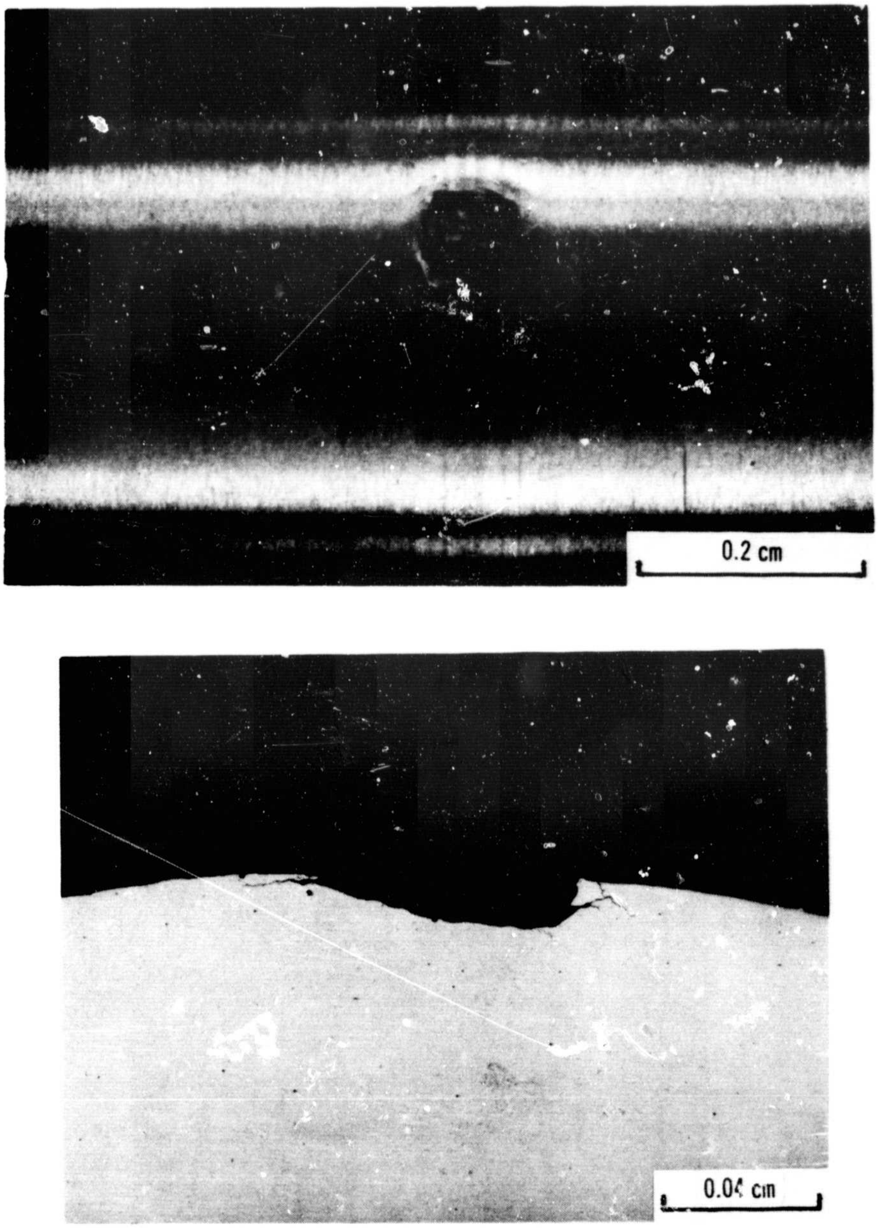


Figure 5. - Typical rolling-element fatigue failure in CBS 600.

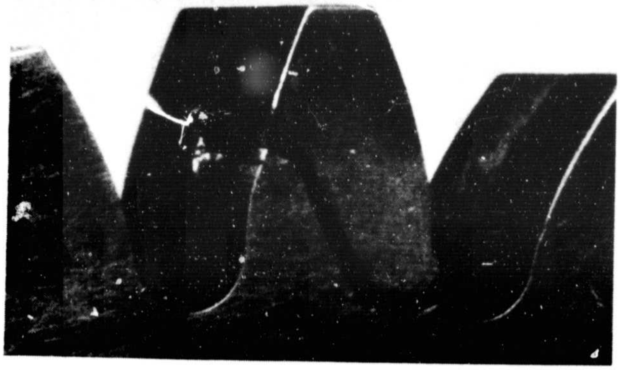


Figure 6. - Typical tooth fracture through a fatigue spall.

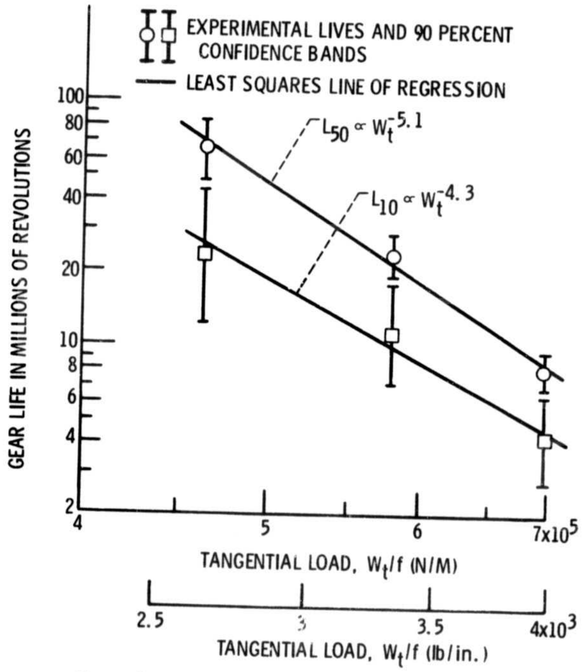
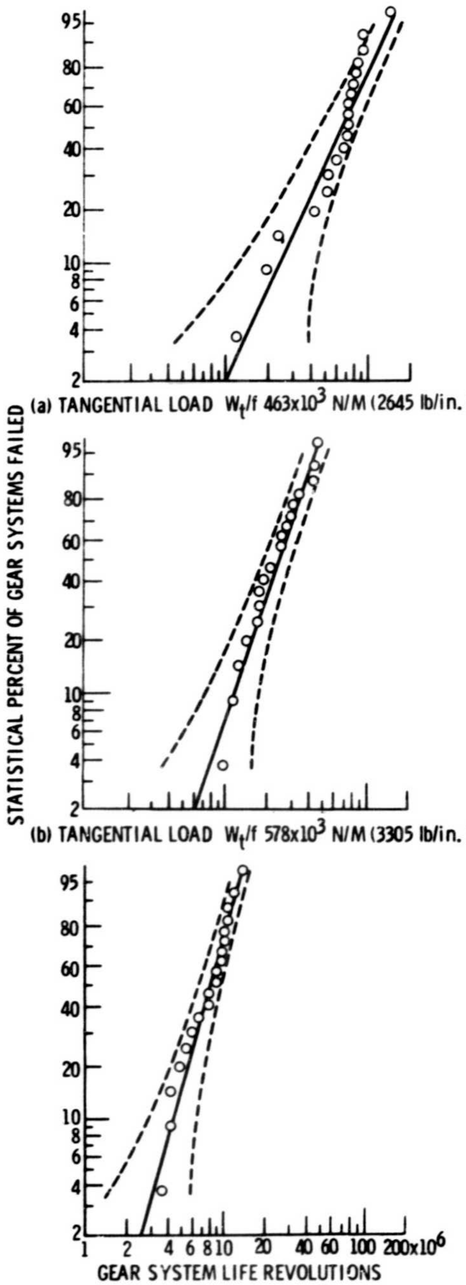


Figure 7. - Load life relationship for (CVM) AISI 9310 steel spur gears speed 10 000 rpm, lubricant superrefined naphthenic mineral oil with additive package.



(a) TANGENTIAL LOAD W_t/f 463x10^3 N/M (2645 lb/in.).

(b) TANGENTIAL LOAD W_t/f 578x10^3 N/M (3305 lb/in.).

(c) TANGENTIAL LOAD W_t/f 694x10^3 N/M (3966 lb/in.).

Figure 8. - Comparison of revised life prediction theory to experimental results for (CVM) AISI 9310 steel spur gears, speed 10 000 rpm, lubricant superrefined naphthenic mineral oil with additive package.

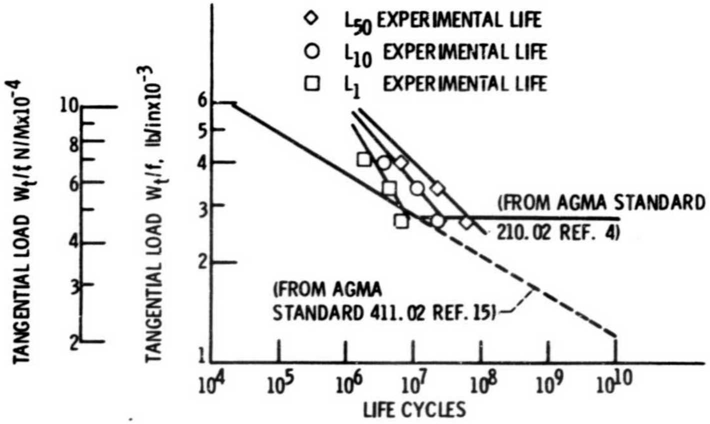
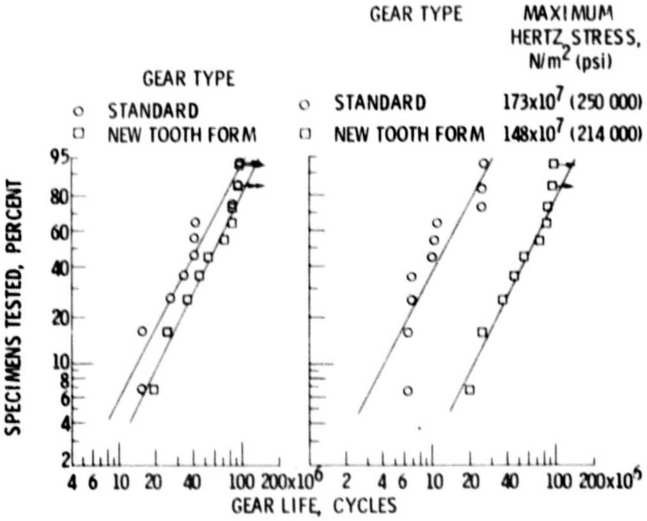


Figure 9. - Comparison of experimental life for (CVM) AISI 9310 spur gears with AGMA life prediction. Speed 10 000 rpm, lubricant superrefined naphthenic mineral oil with additive package.



(a) LIFE AT EQUAL CONTACT STRESSES. MAXIMUM HERTZ STRESS, 148×10^7 N/m^2 (214 000 psi).

(b) LIFE AT EQUAL LOADS. LOAD, 6440 N/cm (3680 lb/in).

Figure 10. - Pitting fatigue lives of standard and new-tooth-form spur gears. Speed, 10 000 rpm; temperature, 370 K (207° F); lubricant, synthetic polyol ester.

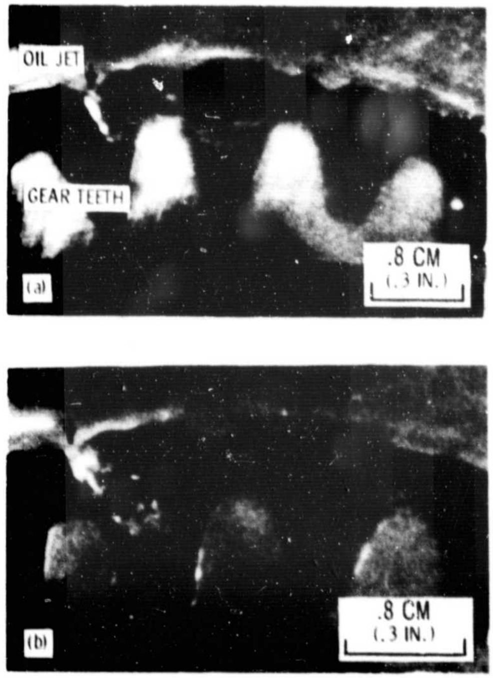


Figure 11. - Oil jet penetrating tooth space and impinging gear tooth; speed, 4920 rpm; oil pressure, 10.5×10^4 N/m^2 (15 psi); xenon and stroboscopic light.

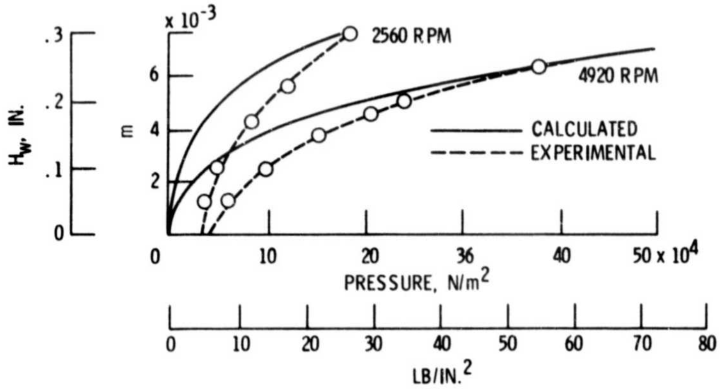


Figure 12. - Calculated and experimental impingement depth versus oil jet pressure at 4920 and 2560 rpm.

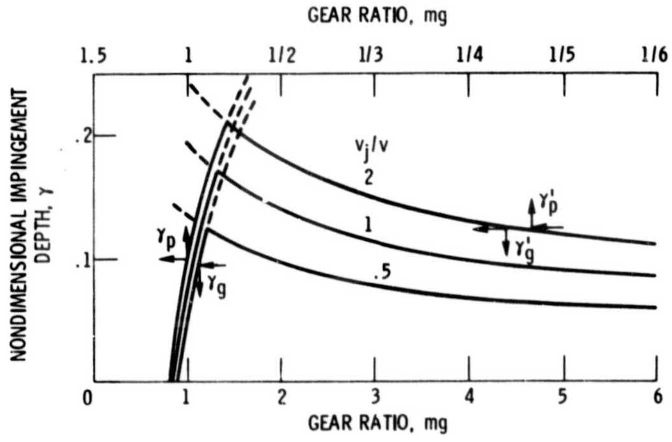


Figure 13. - Gear ratio vs nondimensional impingement depth speed 3600 rpm jet pressure 17×10^4 N/m^2 (25 psi) N_p = 28.

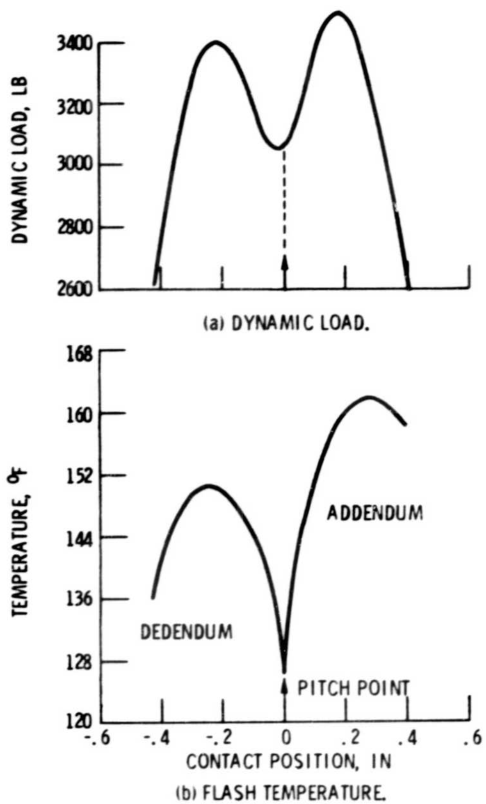


Figure 14. - Computer program results for a spur gear mesh, ratio 1:1, pressure angle 20°, 36 teeth, 2000 lb load, 7800 rpm.

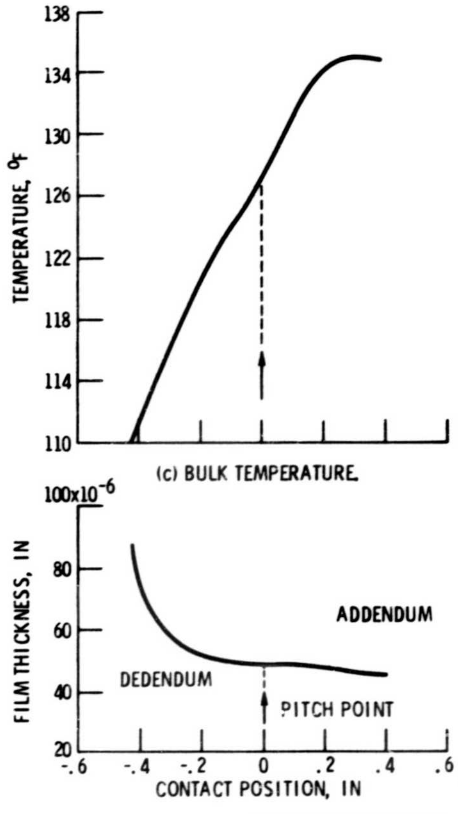


Figure 14. - Concluded.